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Study of a Thermal Aerosol Oil Burner

Honeywell, Inc, Bloomington, Minn

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Industrial Environmental Research
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Research Triangle Park, North Carolina 27711

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September 1977

STUDY OF A THERMAL AEROSOL OIL BURNER

Interagency
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Research and Development
Program Report

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16. ABSTRACT The report gives results of a study of a thermal aerosol oil burner, aimed at counteracting the poor atomization and excess burner capacity that are known to reduce seasonal efficiency and contribute to excess emissions in residential oil burners. Generation of a thermal aerosol of the fuel was shown to improve combustion in terms of: (1) increased quantity and volume of fuel droplets 1 micrometer and smaller; (2) permitted firing rate reduction in standard nozzles of 50 to 70%; (3) when combined with swirl (swirl parameter = 4.5), gave increased combustion efficiency by permitting operation at 2% flue oxygen with nonluminous flame, zero Bacharach smoke No., no hydrocarbons, and no detectable CO; (4) reduced NOx formation in most cases; (5) reduced emissions during burner start-up; and (6) permitted modulation of firing rate without affecting combustion adversely.

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FOREWORD

The present need to use fossil fuel more efficiently and at the same time reduce pollution from combustion processes is placing new demands on residential heating systems. Residential heating systems are one of the major consumers of refined heating oil and gas. They also contribute to atmospheric pollution. The Industiral Environmental Research Laboratory of the U. S. Environmental Protection Agency is concerned with both

- Reducing pollution, and
- Improving efficiency of combustion systems.

This report presents the results of a research project undertaken to explore the idea of using a thermal aerosol generator as an oil burner. It was postulated that the improved atomization would produce clean efficient combustion at relatively low firing rates.

PREFACE

Energy conservation has become a major national concern. At the same time we are committed to improvement of the environment. Ways must be found to reduce waste and pollution in the combustion of fossil fuel for home heating.

It is well known that residential oil burners frequently do not perform at the best efficiency with the lowest possible emission of noxious products. Poor fuel atomization contributes to this problem. The thermal aerosol generator has been used since before World War II to produce smoke from fuel oil. In this device, oil is first vaporized. The vapor then condenses in droplets of around 0.1 micrometer diameter. Droplets of this size produce a dense white fog.

W. L. Tenney (consultant to this project) proposed using a similar principle to atomize fuel oil in an oil burner and obtained a U. S. Patent (4,013,396). His idea was to heat the oil under some pressure and then allow it to be atomized through a nozzle. Some of the oil would flash to vapor and in the process would help break up the liquid phase. The presence of vapor phase in the nozzle also would reduce the flow rate. Reduced firing rate is desirable since most residential oil burners are grossly oversized and this leads to reduced seasonal system efficiency.

This project was undertaken to explore the feasibility of this idea. Since the project was quite modest, it was not possible to study each of the effects in depth. Rather, a survey of conditions was made to assess the promise of this principle and to define areas needing further study.

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ACKNOWLEDGMENTS

We are grateful to W. L. Tenney for his cooperation and stimulating discussions in the application of thermal aerosol technology. As a result of initial discussions, R. H. Torborg, and A. E. Johsnon, under the direction of U. Bonne, carried out an experiment which demonstrated that the thermal aerosol principle was attractive for low capacity residential oil burners. This experiment led to the proposal for this project. We are grateful to EPA for supporting the study.

SECTION 1

INTRODUCTION

Recent studies (1, 2, 3) have shown that excess capacity in residential heating systems leads to excessive stack losses and reduced seasonal system efficiency. A study of 26 oil burners in the Boston area (4) revealed an average excess capacity of 147% (i. e. 247% of design load). The capacity of these burners was then reduced an average of 27. 1% by installing smaller nozzles. It was found, however, that excess air had to be increased an average of 36. 9% to prevent smoke. The increased losses associated with increased excess air largely cancelled the improvement to be expected from reduced capacity.

Many conventional oil burners are too large to heat a residence efficiently. Greater use of insulation compounds the oversize problem. There appears to be a real need for a burner with a capacity of 0. 4 to 0. 6 gph (0. 42 to 0. 53 cm^3/s). Pressure atomizing nozzles of this size have such small orifices and passages that they are prone to becoming plugged after unacceptably short operating times.

It is well known that improved atomization, i. e. smaller drops, permits better mixing of the air and fuel and reduces the tendency to form soot. It would appear, therefore, that a burner that could achieve good atomization without resorting to very small orifices in the nozzle for a flow rate of about 0. 4 cm^3/s should be very attractive.

A droplet of fluid is held together by surface tension. The surface to volume ratio increases when a given volume of fluid is divided into a larger number of smaller drops. Energy input is needed to overcome the surface tension force when the surface area is increased. It is shown in the Appendix that the radius of a drop is given by the following equation:

$$r = \frac{3\sigma}{\Delta E/V}$$

where: σ = surface tension
ΔE = energy input
V = fluid volume

Appendix equation (6) gives the average drop size consistent with energy consideration. Any spray, however, will have drops with a wide range of sizes. Thus while it may not be possible to compute actual drop size from equation (6), it does show the benefits of heated fuel. Heating the fuel both reduces the surface tension and increases the energy content. W. Tenney patented a "Fuel Aerosolization Apparatus" based on a thermal principle (5). In this device fuel oil under pressure is heated and then allowed to expand through a nozzle. The hot oil flashes to a vapor as the pressure drops in passing through the nozzle. Upon cooling after passing through the nozzle, the vapor condenses into very small (less than 1 micrometer) drops which produce a dense white smoke. The U. S. Navy used this device to produce smoke screens in World War II. We proposed this principle as a means of atomizing fuel oil in a residential oil burner. This contract was awarded as a result of that proposal.

The objective of this study was to show that a thermal aerosol generator could be used as an oil burner with a capacity of about 0.4 cm^3/s (0.39 gph) and to define the operating parameters needed for clean, efficient combustion.

SECTION 2

CONCLUSIONS

The use of heated fuel oil to produce a thermal aerosol makes possible the atomization of fuel oil at lower pressures and reduced firing rates. Heating the fuel oil, both No. 1 and No. 2, to 150°C greatly increases the number of droplets in the 0.1 micrometer range.

The formation of bubbles as fuel flows through the swirl passages in the nozzle restricts flow. This permitted standard 0.89 to 1.31 cm^3/s (0.85 to 1.25 gph) hollow cone spray nozzles to be fired at only 0.42 cm^3/s (0.4 gph). Atomization achieved in this manner reduced smoke, hydrocarbons and carbon monoxide.

When combined with air swirl (swirl parameter as defined in the Appendix) equal to 4.5, non-luminous flames were achieved at excess oxygen levels of 2.0% (10% excess air). The Bacharach smoke was zero and there was no measurable hydrocarbon or carbon monoxide. Luminous flames were present without swirl but the thermal aerosol was beneficial in reducing emissions.

The thermal aerosol combined with swirl reduced NO_x formation in most cases when clean burning was achieved. This effect tended to disappear when the excess oxygen was reduced below 2% (10% excess air).

The use of heated fuel improves combustion during burner start-up. Fuel heated to 150°C achieved zero smoke in less than 1.5 minutes. Fuel at 80°C produced a No. 5 smoke at 1.5 minutes but achieved a zero smoke within 5 minutes.

The flow rate of a 1.05 cm^3/s (1.0 gph) nozzle was varied from 0.2 to 0.6 cm^3/s by varying fuel temperature. Clean combustion with low excess oxygen was achieved at all firing rates.

Thus the combination of the thermal aerosol generator with substantial air swirl gives a non-luminous flame with low emissions at an excess oxygen level of 2%. If the flue temperature rise is held to 300°C this results in a combustion efficiency of 81%.

SECTION 3

RECOMMENDATIONS

The results of these experiments show promise for a residential oil burner of 0.41 to 0.52 cm^3/s (0.4 to 0.5 gph) rated capacity. A prototype burner of this size should be built. The burner should have a blower with sufficient pressure rise to supply the vortex mixing chamber. Provision should be made for regeneratively heating the fuel, but a small electric powered heater should be provided for initial heating of the fuel. Excess air should be held in the 5 to 10% range. Also, recirculation should be investigated as a means of further reducing NO_x.

SECTION 4

EXPERIMENTAL PROCEDURES

APPROACH

Our approach was to design a burner that would permit us to control each of the operating variables separately. These were fuel temperature, fuel pressure, fuel flow rate and air flow rate. Since fuel flow is a function of fuel pressure and temperature at the higher fuel temperatures, our first approach was to design a variable area nozzle which would permit independent control of fuel flow rate. It became clear, after two attempts, that the design of a satisfactory variable area nozzle would consume too much time and effort. Instead, we substituted three standard oil burner nozzles. These were rated at 0.89, 1.05 and 1.31 cm^3/s (0.85, 1.0 and 1.25 gph). Using the highest fuel pressures of interest with the small nozzle and the lower pressures with the large nozzle, it was possible to stay within the flow range of interest.

Combustion efficiency was computed from measurement of the oxygen content and temperature of the flue gases. The heat exchanger coupled to the combustion chamber was not designed for efficient energy absorption but rather to simulate the temperature quenching that normally occurs. Therefore, efficiency calculations are based on an assumed flue gas temperature rise of 300°C (540°F).

The flue gas analysis was made with instruments discussed under the section on instrumentation.

Burner Design

The burner consisted of a steel pipe 178 mm (7 in.) i.d. by 508 mm (20 in.) long. The first half was insulated with 12.6 mm (0.5 in.) thick moldable ceramic fiber insulation. * The remainder of the chamber including the inlet end was insulated with 6.35 mm (0.25 in.) of the same material. These dimensions gave a volumetric heat release rate of 1,928 $kJ/s\text{-}m^3$ (186,000 $BTU/hr\text{-}ft^3$). This was consistent with standard practice.

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* WRP Ceramic fiber insulation, Refractory Products Co., 12W. Main St. P.O. Box 309, Carpentersville, IL 60110.

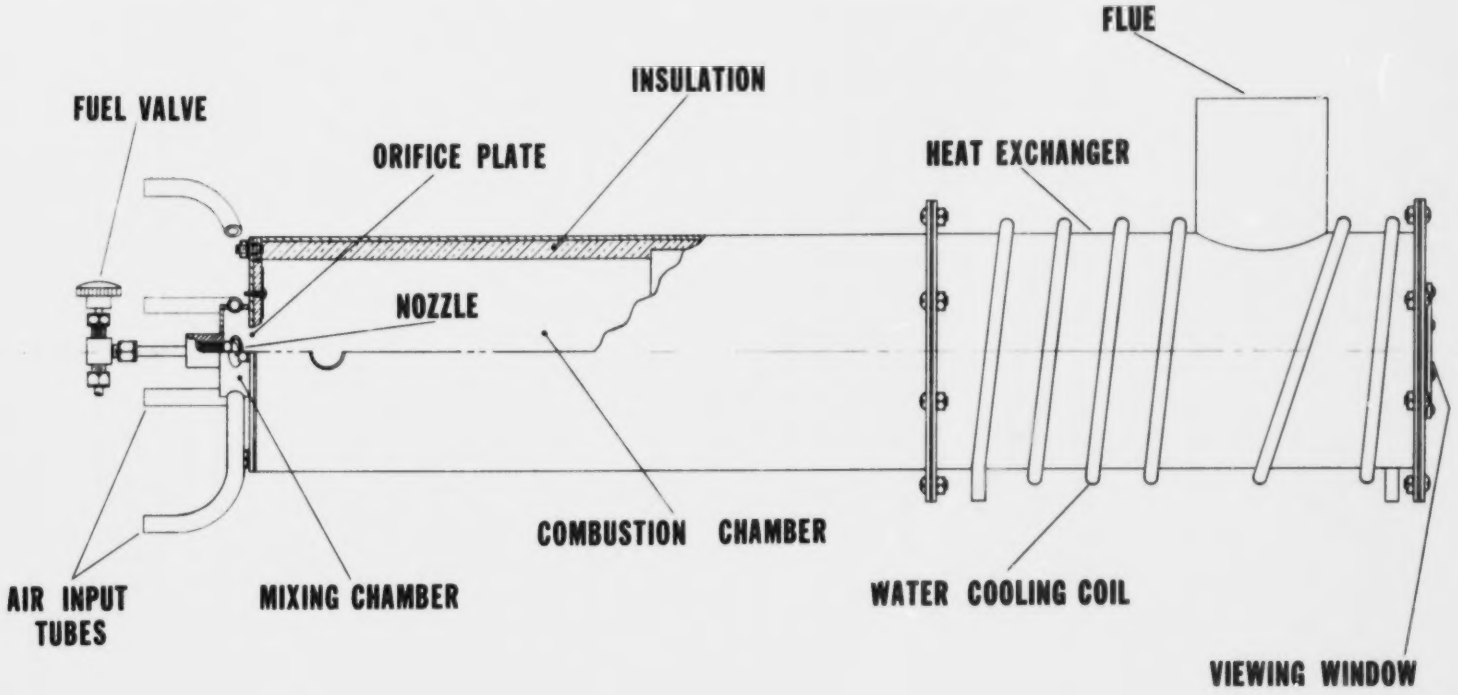

FIGURE 1. EXPERIMENTAL BURNER

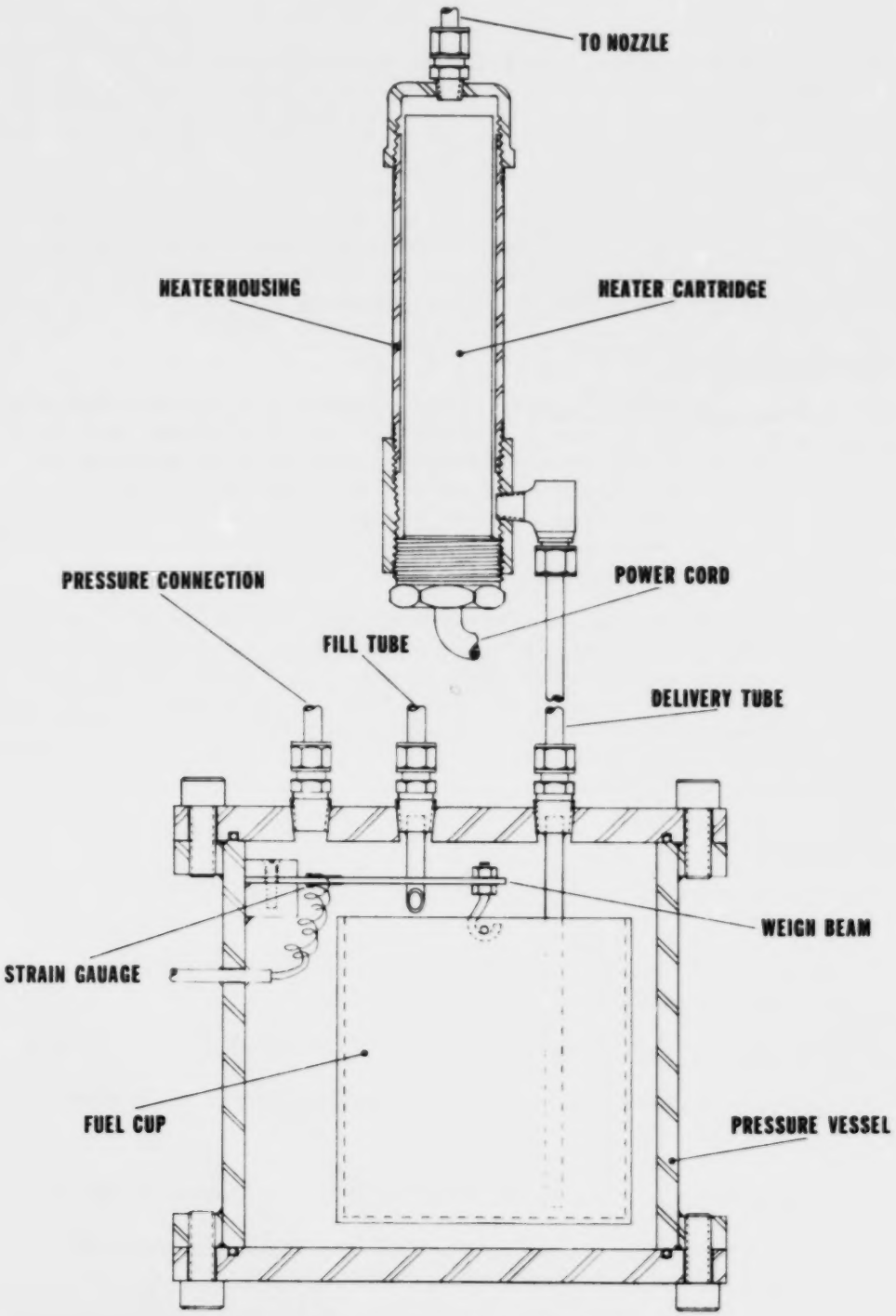

FIGURE 2. FUEL HANDLING SYSTEM

The nozzle was screwed into a fitting attached to the fuel valve. The nozzle in turn sprayed fuel into a vortex mixing chamber attached to the end of the combustion chamber. This vortex chamber was 76 mm (3 in.) dia. by 25 mm (1 in.) long. A 38 mm (1.5 in.) orifice separated the vortex chamber from the combustion chamber.

Air was admitted tangentially to the vortex chamber through four 13 mm (0.5 in.) steel tubes spaced 90^{o} apart around the vortex mixing chamber. The air was supplied through a single orifice meter to a header. The layout of the combustion chamber is shown in Figure 1.

Fuel Handling System

Fuel was supplied from a weigh tank mounted in a pressurized chamber. The weigh tank was mounted on a cantilevered beam equipped with strain gauges. The chamber was pressurized with compressed nitrogen which supplied the atomizing pressure. Fuel flow rate was determined by observing the change in weight of the weigh tank over a measured time interval. The weigh tank had a capacity of 700 cm^3 (0.185 gal) which gave a running time of 20 to 30 minutes before refueling was needed.

The fuel was piped from the weigh tank chamber to the electric powered heater located directly above the weigh tank. This was a 1690 watt heater mounted in a section of 38 mm (1.5 in.) pipe. The high-powered heater was needed to keep the heat flux density at the heater surface to the recommended level of 13,951 W/m^2 (9 $W/in.^2$). The fuel valve and burner nozzle were mounted directly on top of the fuel heater. Layout of the fuel measuring and heating system are shown in Figure 2.

Instrumentation

The instrumentation layout is shown in Figure 3. Instruments used were as follows:

Strain Gauge on Fuel Weigh Tank - Balwin Lima Hamilton SR4 Strain Indicator, Type N.

Oxygen Measurement - Westinghouse, Hagan Oxygen Monitor

Nitrogen Oxides - Aero Chem Chemiluminescence Monitor for NO, NO_x.

Hydrocarbons - Beckman Model 400 Hydrocarbon Analyzer

Carbon Monoxide, Carbon Dioxide, Methane - Honeywell Non-dispersive Infrared Analyzer

Particle Size Measurements - Thermo Systems Inc. Model 3030 Electrical Aerosol Size Analyzer for 0.01 to 1.0 micrometer range and ROYCO Particle Analyzer for the 0.5 to 10 micrometer range

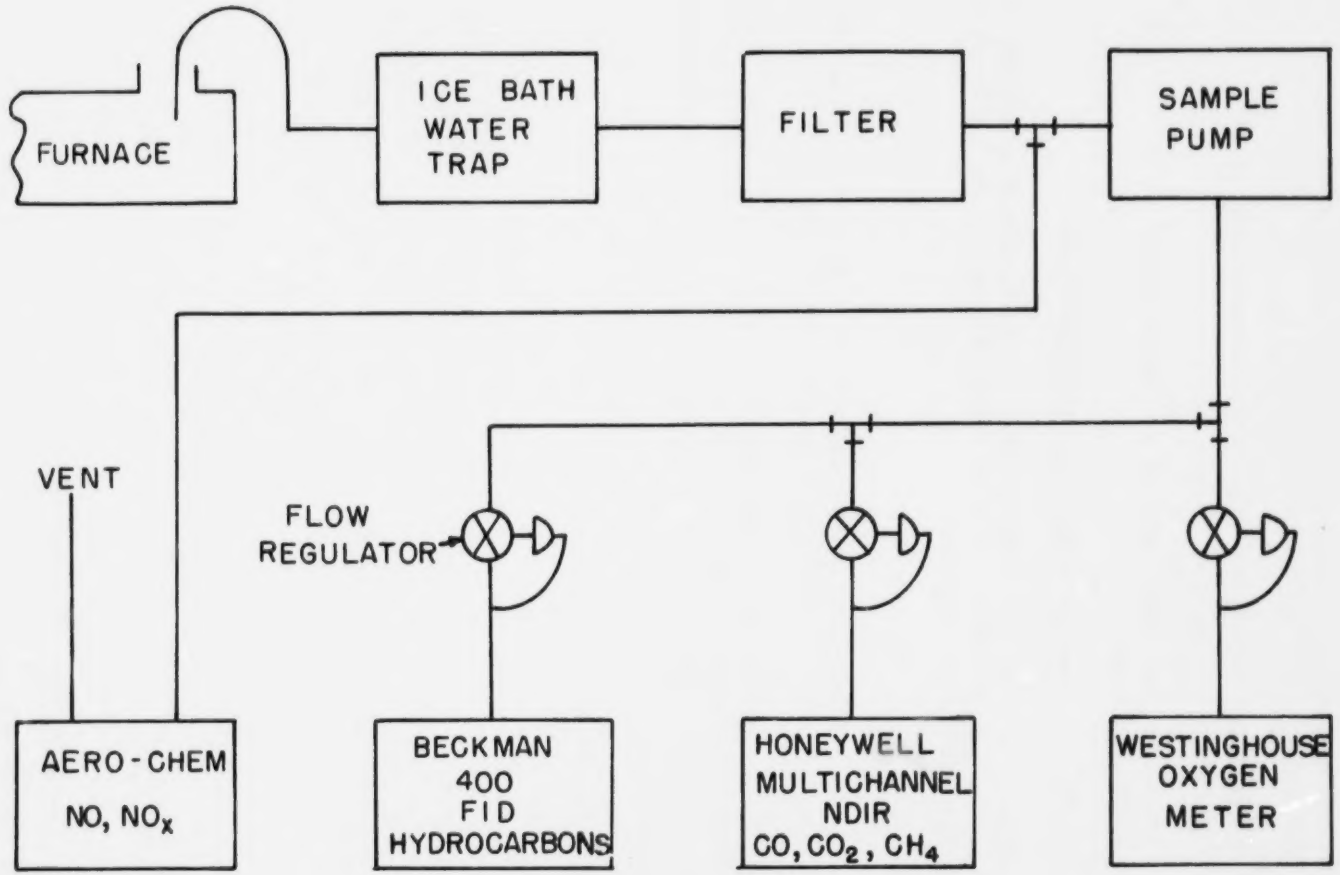

FIGURE 3. INSTRUMENTATION

Fuel Temperature - Copper vs. Constantan thermocouple and Honeywell Class 19 Recorder.

Fuel Pressure - Heise precision pressure gauge, 0-50 PSI range.

Air Flow - 0.5 in. dia. orifice and water filled "U" tube manometer.

Smoke Measurements - Bacharach Smoke Tester

These instruments all had meters which were read and manually recorded. Thus, measurements had to be made during steady state operation or during slowly changing transients.

TEST PROCEDURE

The test procedure was to fill the fuel weigh tank with No. 1 or No. 2 fuel oil. Properties of the fuel oil are presented in Table 1.

TABLE 1. FUEL PROPERTIES
(Measured by Twin City Testing Laboratory)

Property	Value	
	No. 1 Oil	No. 2 Oil
API Gravity, 15.5°C (60°F)	42.0	34.8
Heating Value		
(J/m^3)	538,862	545,661
(BTU/gal)	134,924	136,627
Carbon (%)	86.26	87.15
Hydrogen (%)	13.58	12.68
C/H	6.35	6.87
Sulfur (%)	0.19	0.37
Nitrogen (%)	0.0064	0.012
Nickel, ppm	26	32
Vanadium, ppm	23	24
Lead, ppm	28	42

The desired pressure on the fuel would be set. The fuel valve was opened slightly to bleed air from the system and fill the fuel heater. The valve was then closed and power to the heater was adjusted to give the desired fuel temperature in the heater. Since the nozzle tended to pick up considerable heat from the flame, it was necessary to compensate for this in adjusting the fuel heater temperature. When the desired temperature was reached, the air supply was turned on, fuel was turned on and ignited with a propane torch. After about 2 or 3 minutes the burner insulation was

hot and the flame was stabilized. The air flow was then adjusted to give the desired excess oxygen level. The various instruments were then read. Following this the air flow or fuel pressure was adjusted to a new value and a new set of readings was taken. Usually about four sets of readings were taken before it was necessary to refuel.

The large number of permutations of the variables made it impractical to test all possible combinations. Instead, operating parameters were selected to cover a wide operating range and additional detail when results of special interest were observed. Tables 2 and 3 show the combinations of firing conditions studied. A brief test of the effect of swirl and a transient test were carried out in addition to the steady state tests.

TABLE 2. FIRING CONDITIONS FOR NO. 1 FUEL OIL

Nozzle cm^3/s (GPH)	Fuel Temp °C	Fuel Pressure kPa gage	Excess Oxygen %
0.89 (0.85)	20	207	2; 4; 8
	20	276	2; 4; 8
	20	345	<1; 2; 4; 8
	150	207	2; 4; 8
	150	276	2; 4; 8
1.05 (1.00)	20	138	2; 4; 8
	20	207	2; 4; 8
	20	276	4; 8
1.31 (1.25)	20	103	2; 8
	20	138	2; 8
	20	207	2; 8
	150	103	2; 8
	150	138	2; 8
		207	2; 8

TABLE 3. FIRING CONDITIONS FOR NO. 2 FUEL OIL

Nozzle cm^3/s (GPH)	Fuel Temp °C	Fuel Pressure kPa gage	Excess Oxygen %
0.89 (0.85)	20	207	2; 4; 6; 8
	20	276	<1; 2; 4; 8
	20	345	<1; 2; 4; 8
	90	207	2; 4; 8
	90	276	2; 4; 8
	90	345	<1; 2; 4; 8
	150	207	2; 4; 6; 8
	150	276	2; 4; 8
	150	345	<1; 2; 8
1.05 (1.00)	20	138	2; 4; 8
	20	207	2; 4; 8
	20	276	<1; 2; 4; 8
1.31 (1.25)	20	103	2; 4; 8
	20	138	8
	150	103	2; 4; 8
	150	138	2; 4; 8
	150	207	<1; 2; 4; 8

SECTION 5
RESULTS AND DISCUSSION

The experimental results are presented in Figures 4 through 17 and Tables 4 and 5.

AEROSOL SIZE DISTRIBUTION

Figure 4 shows how the aerosol drop size distribution is affected by fuel temperature and pressure. Figure 5 shows the relative volume distribution as a function of drop size. Fuel temperature of 200°C and above produced substantial numbers of drops in the 0.1 micrometer range. However, the fuel temperature had to be 250°C or higher to produce a large volume concentration.

The data in Figure 5 cannot be used to estimate air/fuel ratio because the measurements ignore droplets bigger than about 5 micrometers. The larger droplets tended to settle on the walls of the sampling tube and sample chamber. While the number of larger drops was small compared with smaller drops, the mass of fluid in the larger drops probably was substantially greater. Figure 5 is useful for comparing the relative volume of fuel in the 0.03 to 3.0 micrometer range.

As expected, increased pressure and lighter weight (No. 1) fuel increased the number of small drops. When converted to a volume or mass distribution the distribution peaks at around 3 micrometers. The apparent discontinuity at around 1.0 micrometer was at least partially due to the use of two different measuring instruments. A Thermo-Systems Inc. model 3030 Electrical Aerosol Size Analyzer was used in the range 0.01 to 1.0 micrometers. This instrument is based on a measurement of the charged particle current carried by the collected particles after they have been charged. A ROYCO Particle Analyzer was used to measure the concentration of drops in a 0.5 to 10 micrometer range. This instrument is based on an optical principle.

These instruments required a sample of the fuel spray to be fed into a separate sampling chamber. An aerosol sample was then fed into the instrument from this chamber. Only the smaller air-borne drops would stay in suspension through this sampling system. Hence the larger drops in the fuel spray were ignored.

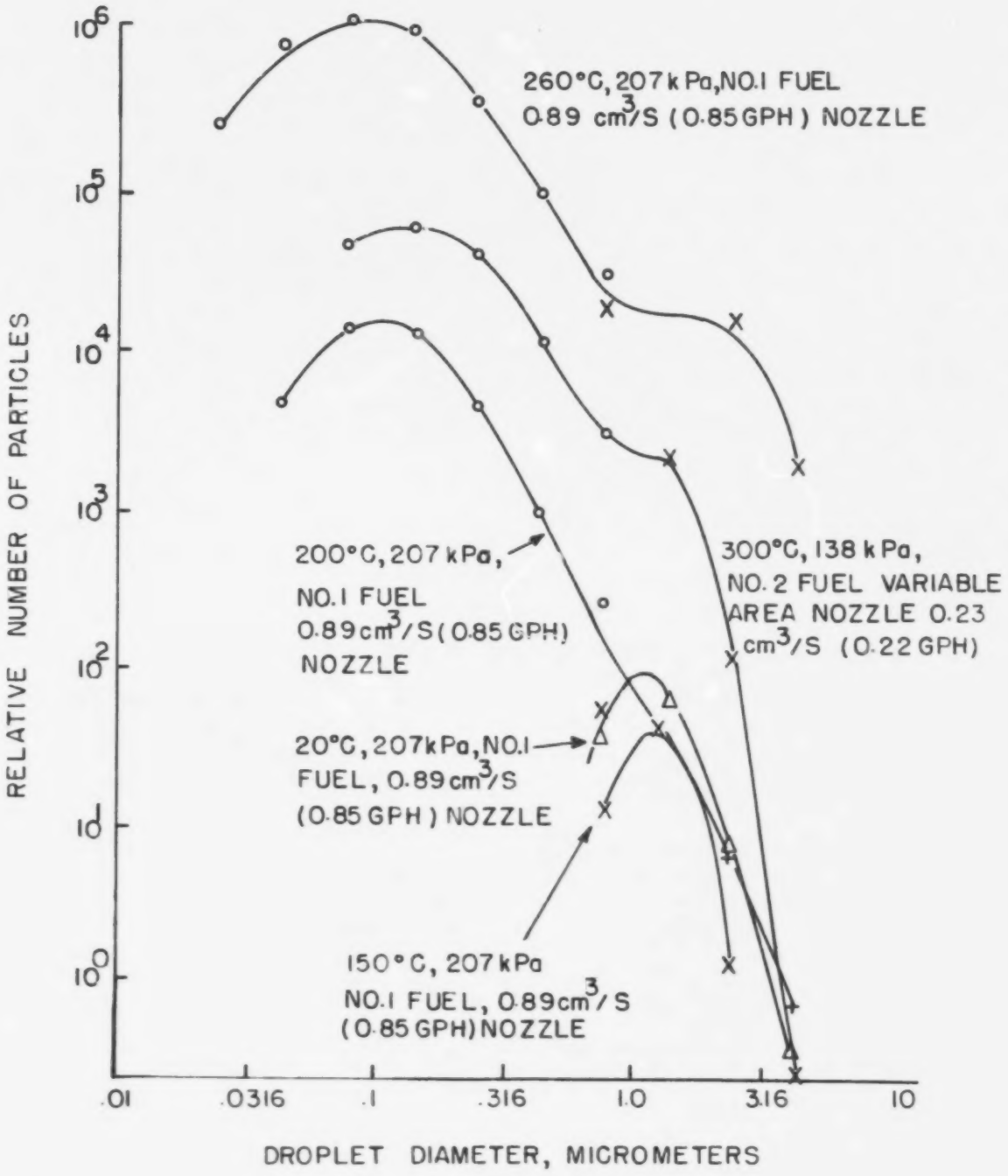

FIG. 4 SPRAY SIZE DISTRIBUTION

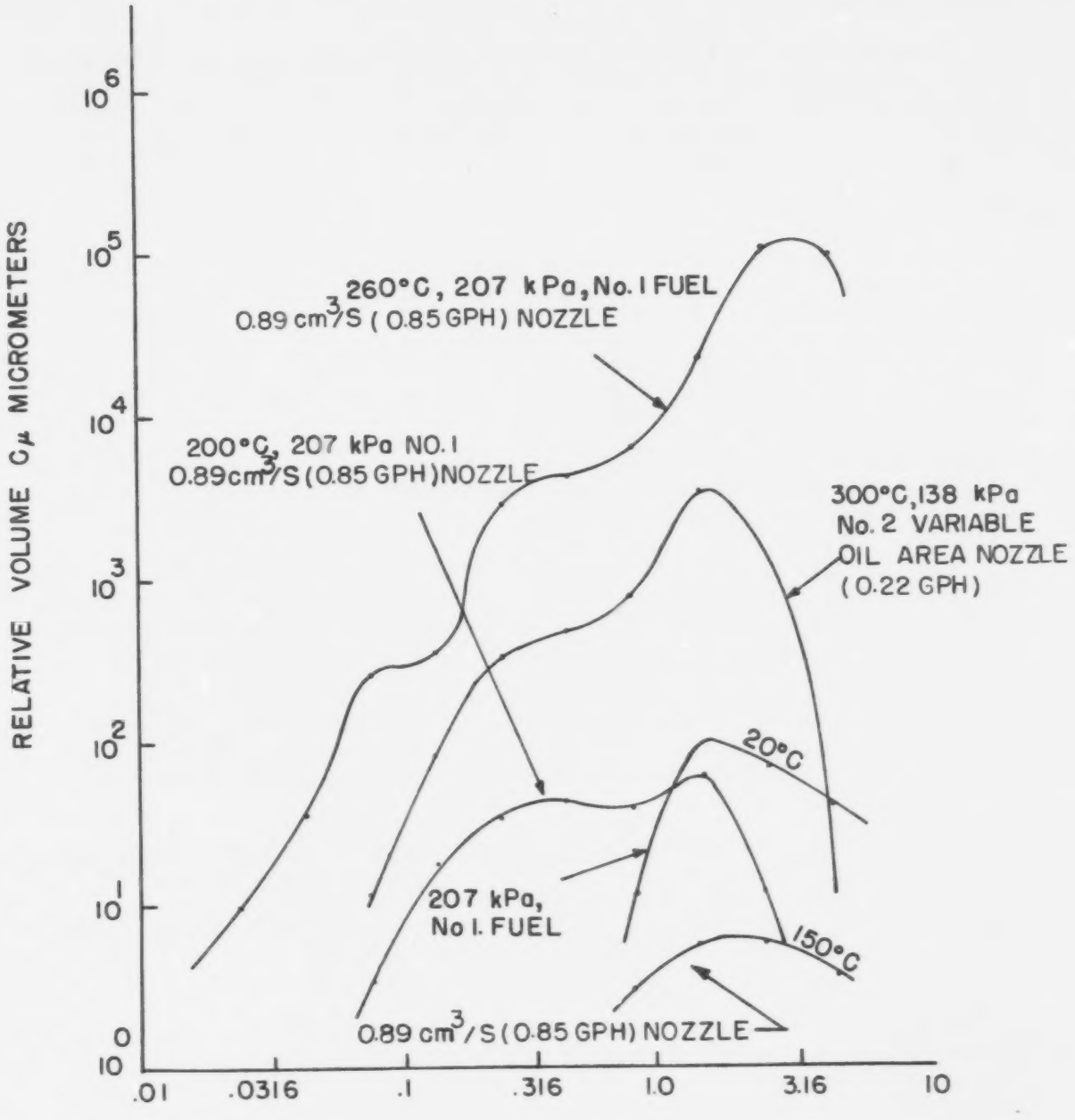

FIGURE 5. SPRAY MASS SIZE DISTRIBUTION

FIRING RATE

Figures 6 through 11 show the effect of fuel temperature and pressure on flow for each of the three nozzles and each fuel. Our objective was to explore conditions in the flow range around 0.42 cm^3/s (0.4 gph). To do this the pressure range was lowered from 207 to 345 kPa (30-50 psi) with the 0.89 cm^3/s (0.85 pgh) nozzle to 103-207 kPa (15-30 psi) the 1.31 cm^3/s (1.25 gph) nozzle. Fuel pressure was very stable and was controlled by the regulated nitrogen pressure in the fuel chamber.

Fuel flow measurements were made with the burner operating. The nozzle picked up considerable heat from the flame. Consequently the nozzle temperature was substantially higher than the fuel heater temperature. The nozzle temperature was measured at the fitting into which the nozzle was screwed. The nozzle orifice temperature probably was a few degrees higher. Because of these complications the nozzle temperature and therefore the fuel temperature varied somewhat as a function of flow rate. We endeavored to compensate for this, but a 10°C uncertainty in the temperature of fuel entering the nozzle probably occurred. Since we were looking for the effect of a 100°C, or higher, fuel temperature change, the uncertainty was not serious.

The flow decreased with decreasing pressure as expected. We expected flow to increase slightly with increasing temperature due to lower viscosity. When the boiling range was reached the flow was expected to decrease rapidly due to vapor formation in the nozzle orifice. However, in all cases except that with No. 1 oil in the 0.89 cm^3/s (0.85 gph) nozzle, flow decreased with increasing temperature. We believe this was due to the fact that fuel oil boils over a rather wide temperature range. The light ends probably produce some small bubbles in the nozzle orifice at fuel temperatures as low as 100°C. The restriction produced by bubble formation increases with increasing fuel temperature.

The one case where the 0.89 cm^3/s (0.85 gph) nozzle was operated at 205-215°C and 276 kPa (40 psi) with No. 1 oil (Figure 6) did show a substantial flow restriction. The No. 1 oil probably produced a substantial amount of vapor under these conditions. No. 2 oil (Figure 7) did not show any break, but probably would have shown a break in the curve at some higher temperature.

In general we found we could operate reliably at 150°C and lower, but coking became a problem during shut down at higher fuel temperatures. Apparently, when the fuel was shut off, heat from the combustion chamber raised the nozzle temperature even higher than its operating temperature.

EFFICIENCY

Combustion efficiency was calculated from excess oxygen concentration and an assumed flue temperature rise. The actual flue temperature rise in any real furnace would be a function of heat exchanger design. We, therefore, assumed that for any given set of combustion conditions, a heat exchanger could be designed to achieve the assumed flue temperature rise. A flue temperature rise of 300°C (540°F) is a reasonable value.

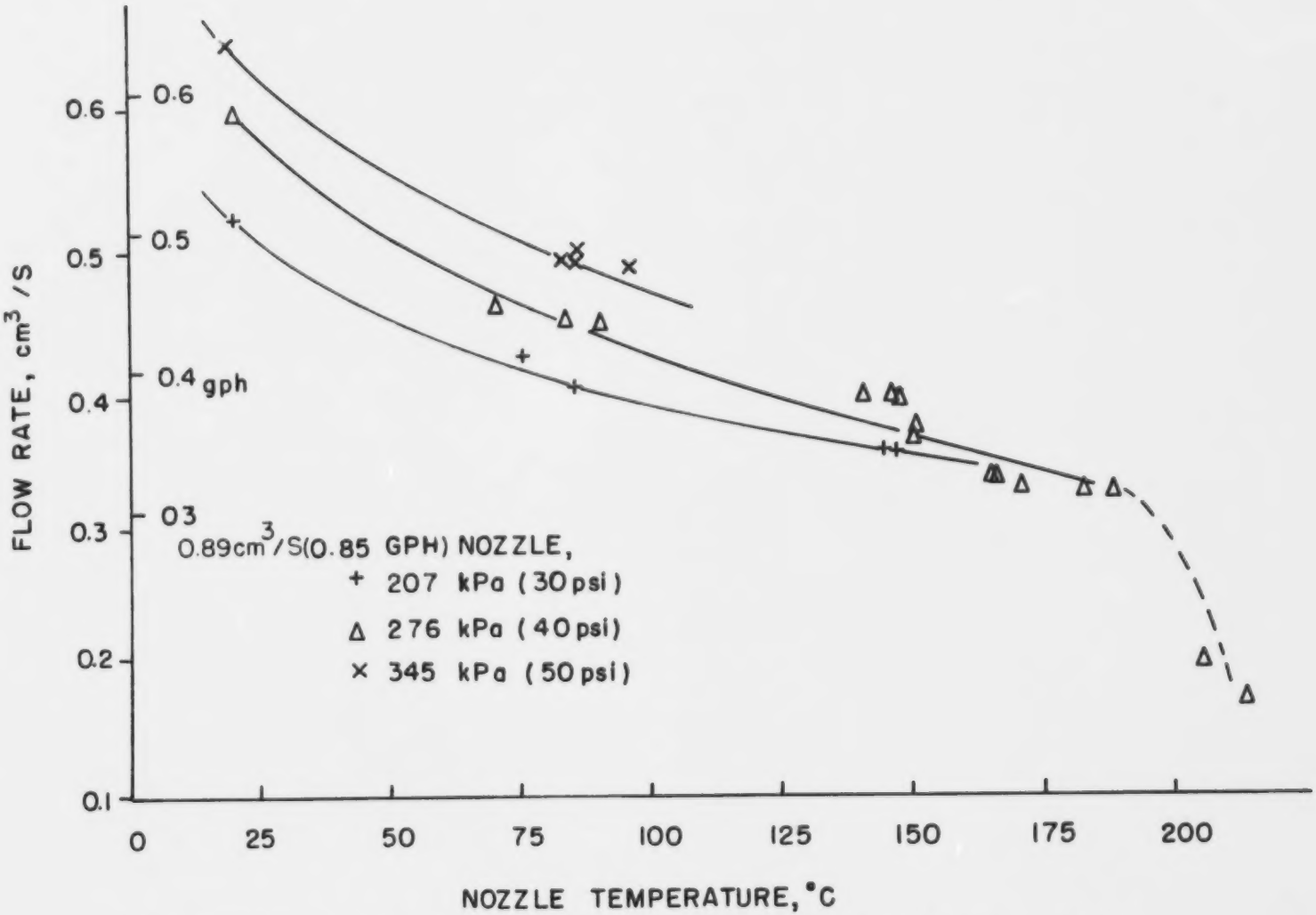

FIG.6 EFFECT OF FUEL TEMPERATURE AND PRESSURE ON FLOW WITH NO.1 OIL

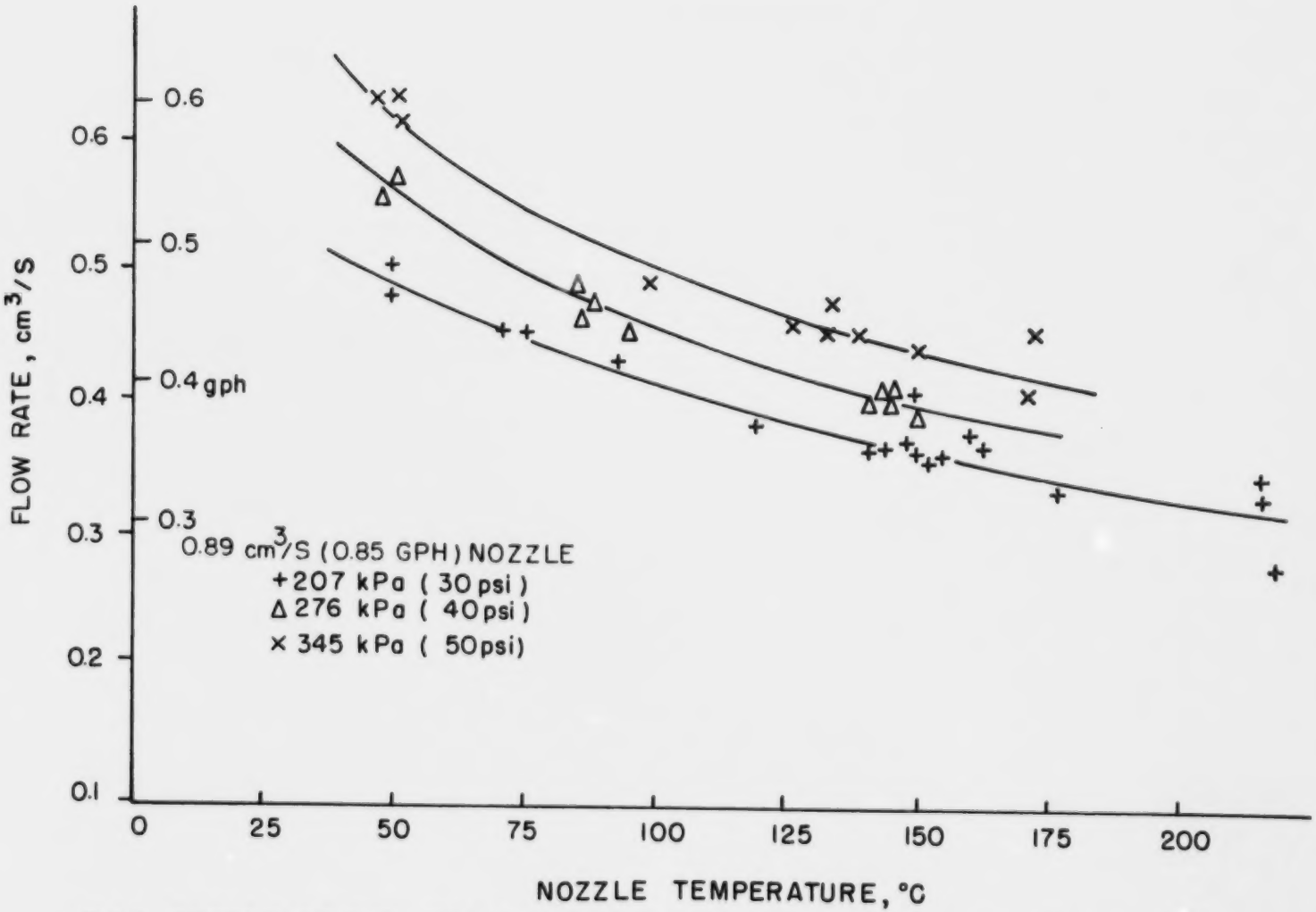

FIG.7 EFFECT OF FUEL TEMPERATURE AND PRESSURE ON FLOW WITH NO. 2 OIL

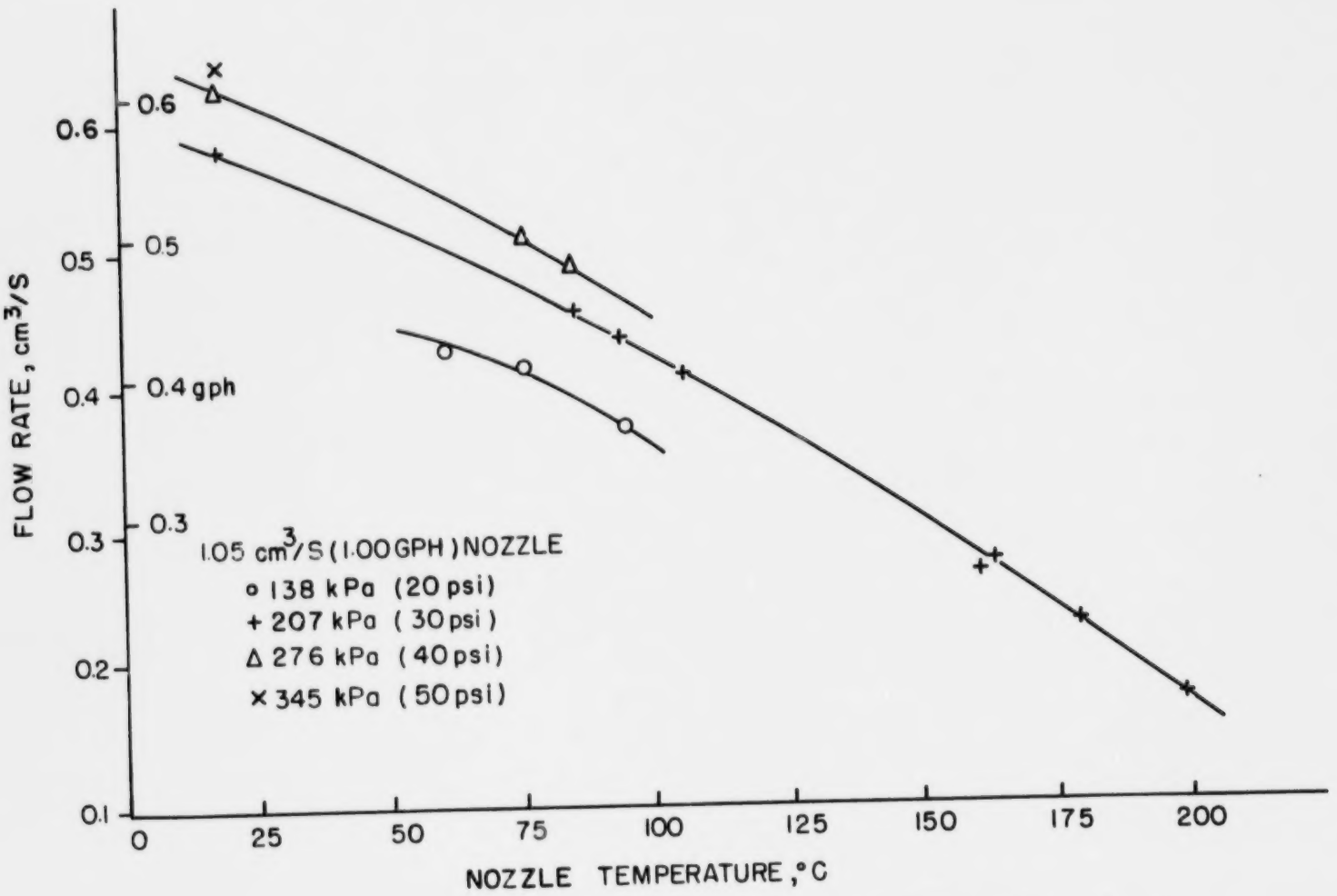

FIG. 8 EFFECT OF FUEL TEMPERATURE AND PRESSURE ON FLOW WITH NO.1 OIL

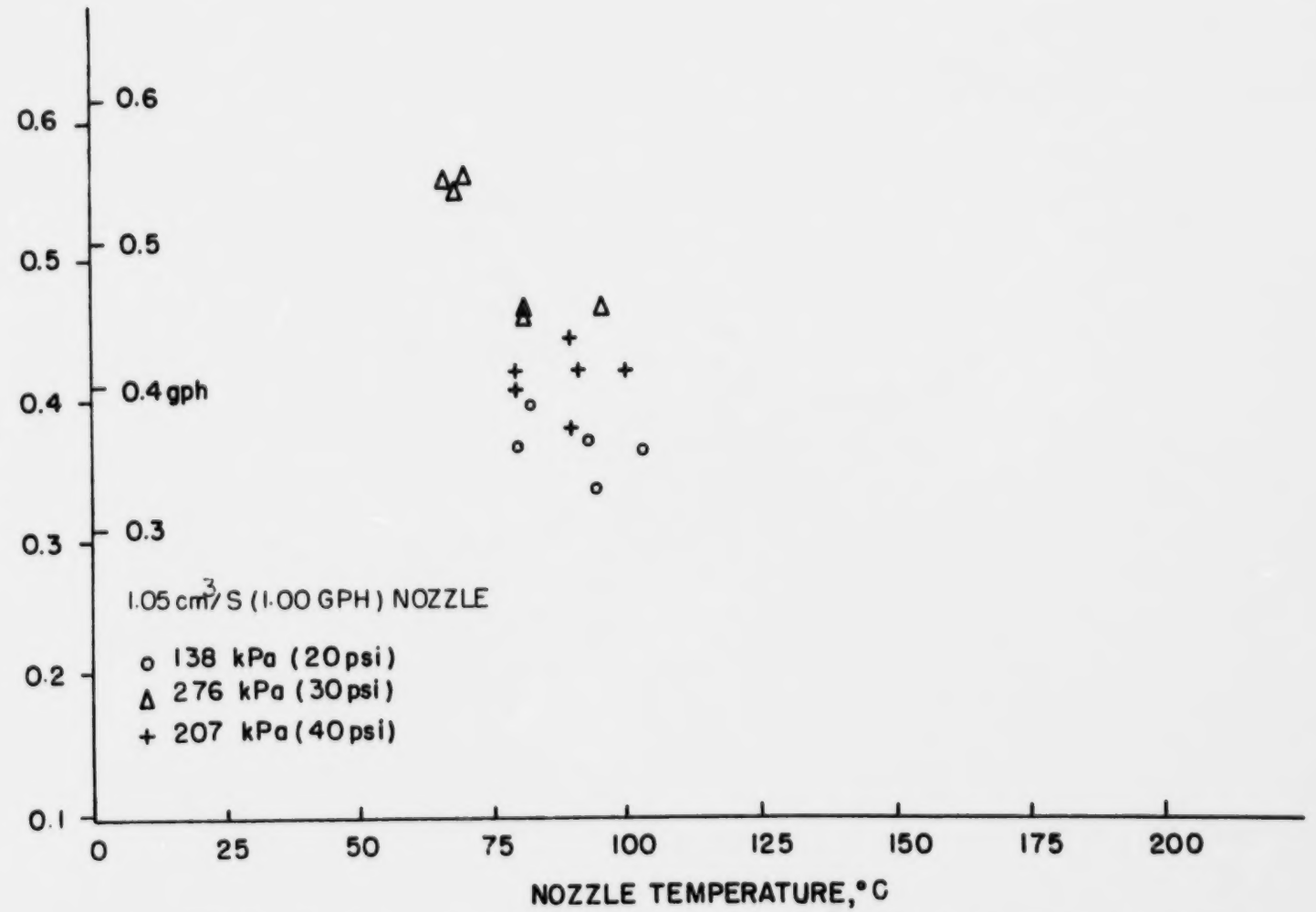

FIG 9 EFFECT OF FUEL TEMPERATURE AND PRESSURE ON FLOW WITH NO. 2 OIL

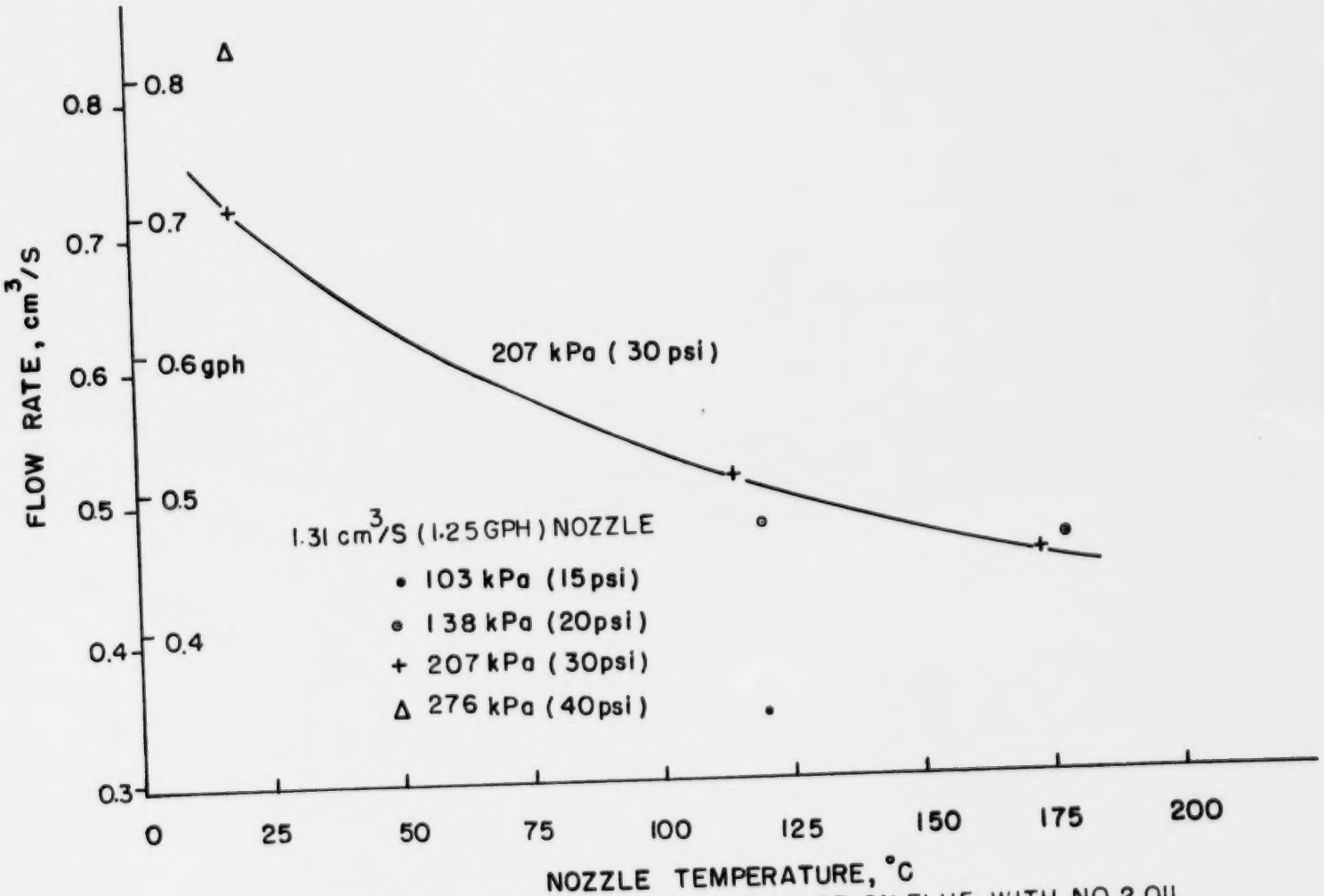

FIG. 10 EFFECT OF FUEL TEMPERATURE AND PRESSURE ON FLUE WITH NO 2 OIL

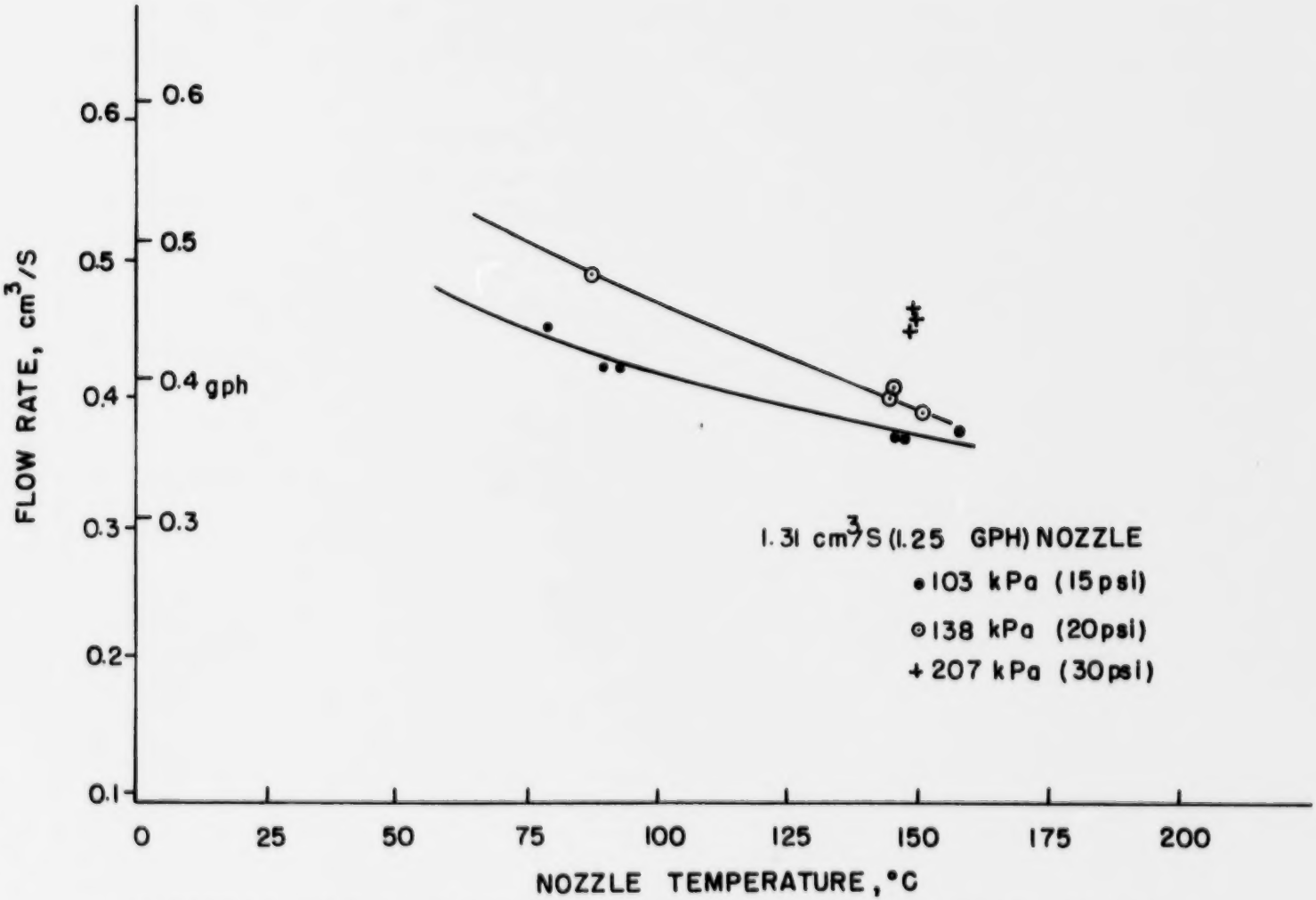

FIG 11 EFFECT OF FUEL TEMPERTURE AND PRESSURE ON FLOW WITH NO.1 OIL

Figure 12 presents a correlation among excess oxygen, flue carbon dioxide and excess air for the two fuels studied. A 0°C (32°F) dew point was used in preparing Figure 12 because we used an ice bath water trap to remove moisture from the flue sample.

Figure 13 gives the gross steady state efficiency (based on higher heating value of fuel) as a function of flue temperature and oxygen content.

Swirl in combustion chambers has been defined by different authors in somewhat different ways (6). We have used a simplified form of Beer's definition which is derived in the Appendix. The swirl parameter so defined for this combustion chamber was 4.5. It probably would be considered moderately high.

This swirl produced clean combustion under almost all firing conditions. It was feasible, therefore, to assume a burner of this type could be operated at 2% excess oxygen (10% excess air). This would give a steady state efficiency of slightly over 81% with a 300°C flue temperature rise. A heat exchanger designed to achieve a lower flue temperature rise obviously would raise the efficiency.

EMISSIONS - STEADY STATE

Nitrogen Oxides

Although the nitrogen oxides are reported as NO_x, a cold trap used in the sampling line (Figure 3) probably removed most of the NO_2 and some hydrocarbons. It was necessary to add this cold trap to protect the NDIR Instrument and Oxygen Analyzer from any condensed water droplets. The experimental set-up dictated the location at the point shown in Figure 3. Under most operating conditions, the concentration of condensed species was probably quite small.

The response of the gas analyzing instruments and sampling system was quite slow. Thus, only steady state or slowly changing conditions could be observed. With the exception of one test, the data reported are steady state values.

The oxidation of nitrogen in a flame depends on time and temperature. Higher temperatures increase the rate of formation of NO_x, but shorter residence times reduces the production of NO_x. Figures 14 and 15 show both of these effects for the 0.89 cm^3/s (0.85 gph) and 1.31 cm^3/s (1.25 gph) nozzles operating with No. 2 fuel oil.

The flue NO_x measured, increased as the excess oxygen decreased due to the higher flame temperature associated with more nearly stiochiometric combustion. Incomplete mixing limited this effect at an excess oxygen level of about 0.3% with the 0.89 cm^3/s (0.85 gph) nozzle when operated at 344 kPa (50 psi) with 50°C, No. 2 oil. The flow rate under these conditions was 0.49 cm^3/s (0.47 gph) or 55% of rated capacity. The 1.31 cm^3/s (1.25 gph) nozzle operated at 207 kPa (30 psi) with No. 2 oil at 150°C had a flow rate of only 0.45 cm^3/s (0.43 gph) or 34% of capacity. It showed the effect of incomplete mixing (maxima in the NO_x curve) at about 1% excess oxygen.

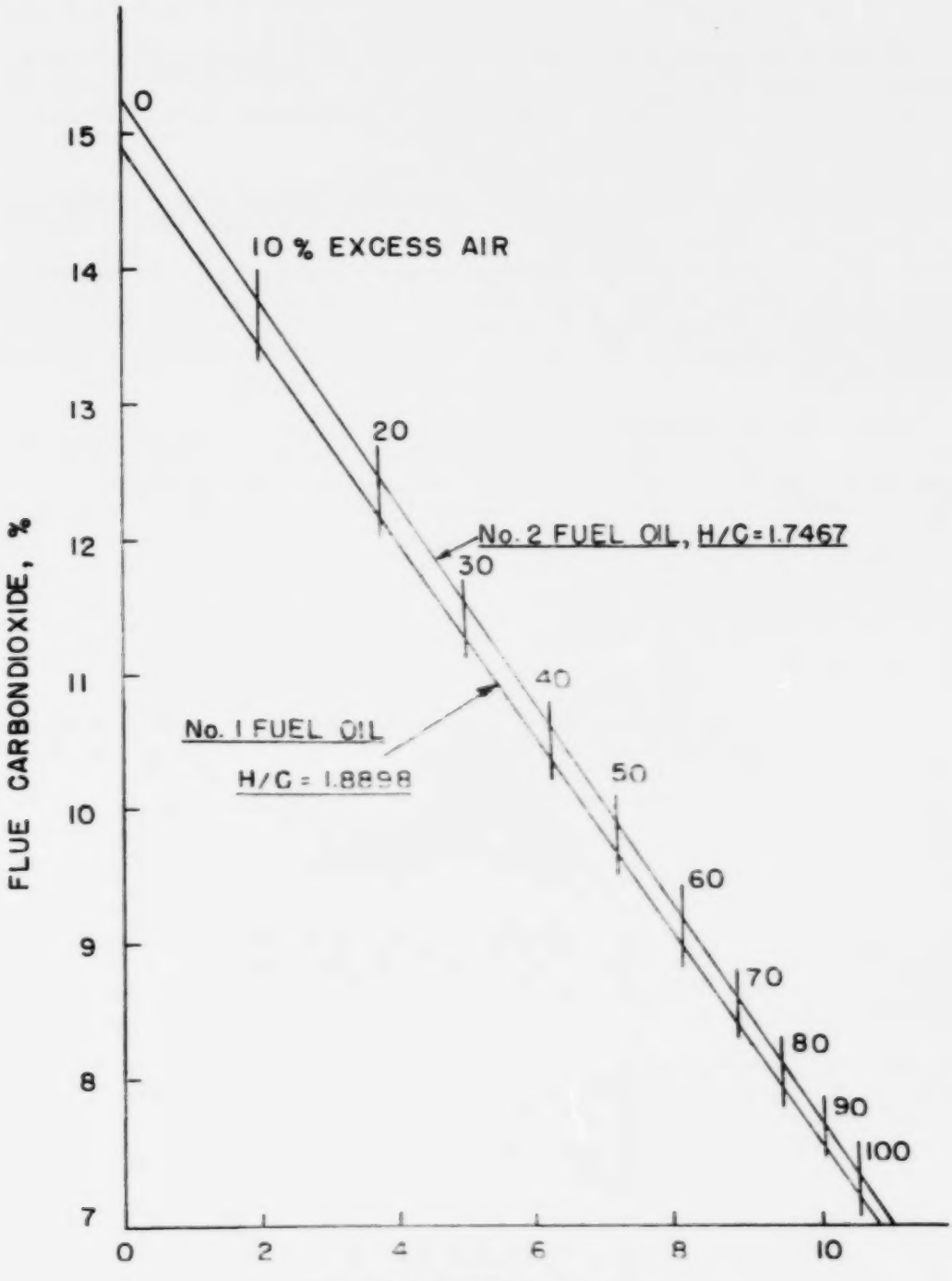

FIGURE 12. CORRELATION OF FLUE PARAMETERS

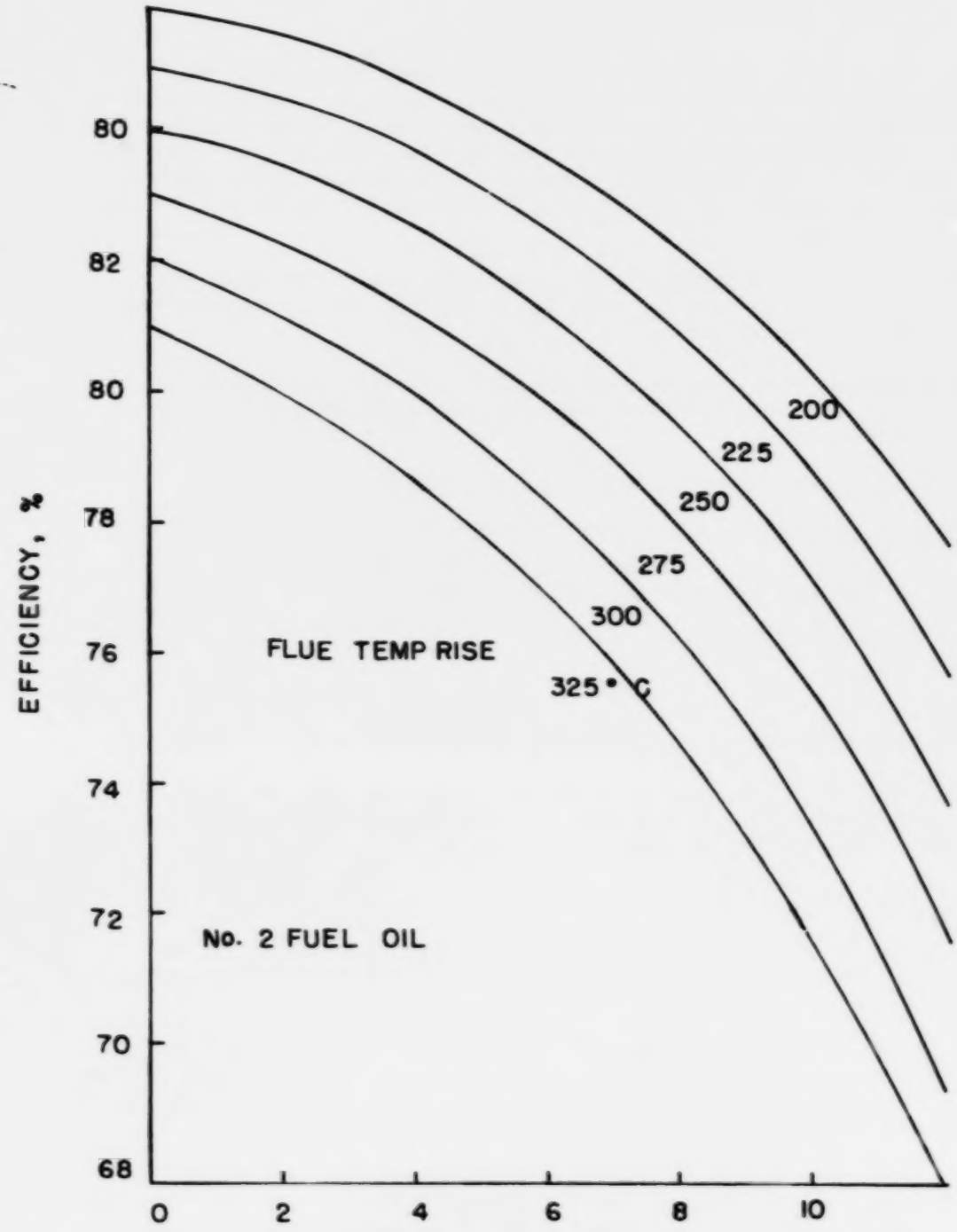

FIGURE 13. GROSS STEADY STATE EFFICIENCY

In most, but not all, cases the NO_x decreased with increasing fuel temperature and pressure. Improved atomization achieved with increased fuel temperature and pressure would permit faster mixing in the turbulent air stream. This, in turn, would lead to smaller flame volume and less residence time in the high temperature region. Measurements and calculations by other investigators (6, 7) also have shown that increased swirl tends to reduce NO_x formation.

Thus the use of high turbulence and a thermally augmented atomizer appears to be beneficial in reducing NO_x formation. Provision for recirculation could enchance this situation.

The fuel nitrogen was measured as shown in Table 1. If this was all converted to NO_x, the contribution of fuel nitrogen to the NO_x concentration is given in Table 4.

TABLE 4. CONTRIBUTION OF FUEL NITROGEN TO NO_x

Fuel No.	Fuel Nitrogen %	NO_x Normalized to 3% Flue 0_2 PPM
1	.0064	8.1
2	.0120	15.5

Figures 14 and 15 show that at 8% excess oxygen the NO_x can be attributed mainly to fuel nitrogen. However, at lower flue oxygen levels, more of the NO_x must come from atmospheric nitrogen.

Other Emissions

Smoke, total hydrocarbons and carbon monoxide were essentially undetectable for all firing conditions except when excess oxygen levels were well below 1%.

The total hydrocarbons analyzer had two ranges, 100 ppm and 1000 ppm full scale. The minimum detectability was about 1 ppm. The carbon monoxide sensor was calibrated for 1000 ppm full scale with a resolution of about 10 ppm. Methane could be detected at a level of about 5 ppm.

The high degree of swirl achieved complete combustion of the fuel with almost no excess air. During one run with the 0.89 cm^3/s (0.85 gph) nozzle firing No. 2 oil at a nozzle temperature of 165°C, the excess oxygen was adjusted to less than 1% with 0 to 1 smoke number. Total hydrocarbons, carbon monoxide, and methane were unmeasurable. The NO_x concentration was then 55 ppm. The inlet air pressure was then reduced slightly to yield a 0.15% decrease in excess oxygen (0.2% increase in flue CO_2). Carbon-monoxide increased to more than 1000 ppm and NO_x decreased to 51 ppm. Total hydrocarbons and methane were still unmeasurable, however. Thus the point of incomplete combustion was very pronounced and occurred at about 0.8% excess oxygen.

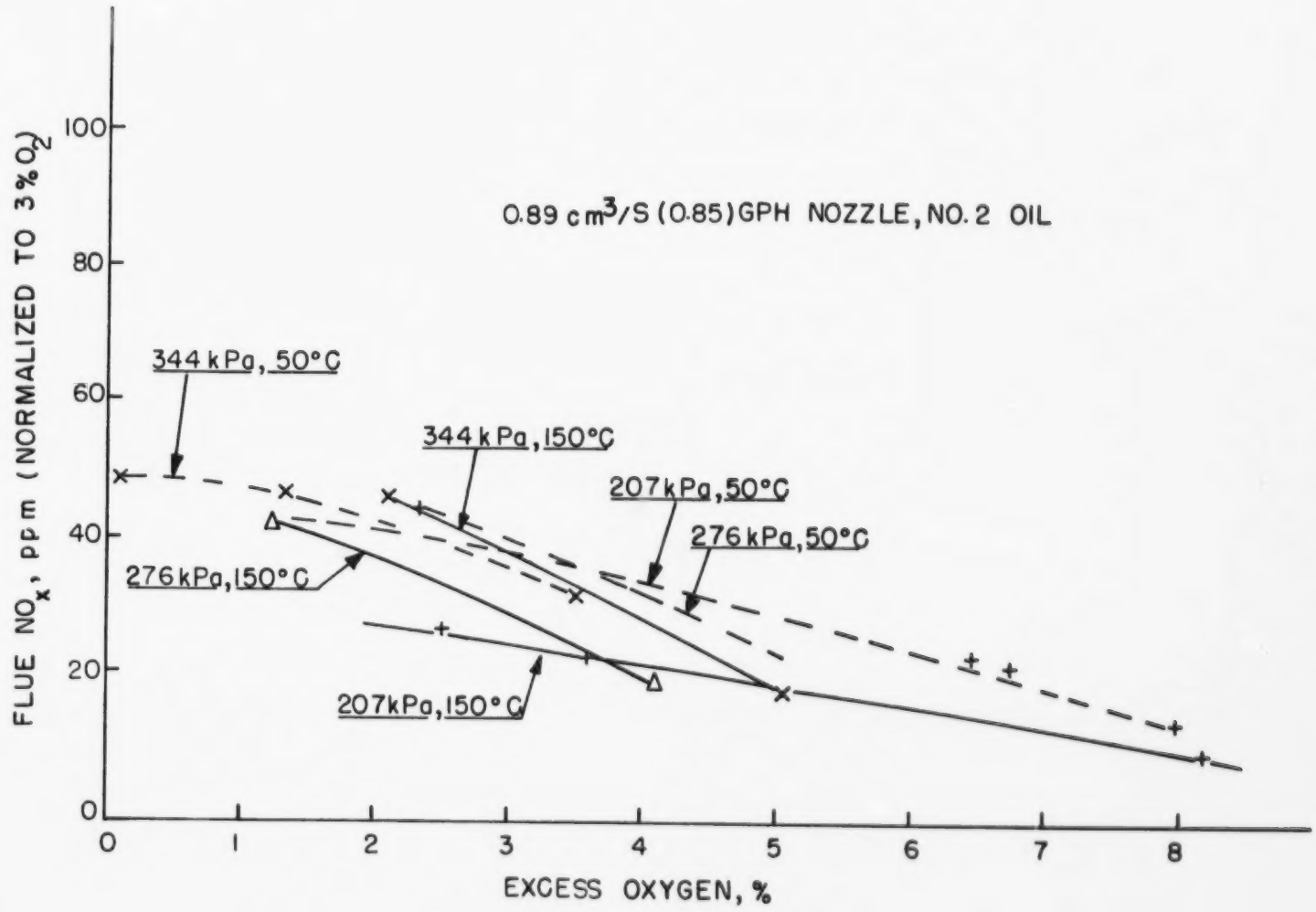

FIGURE 14. EFFECT OF OPERATING PARAMETERS ON NO_x

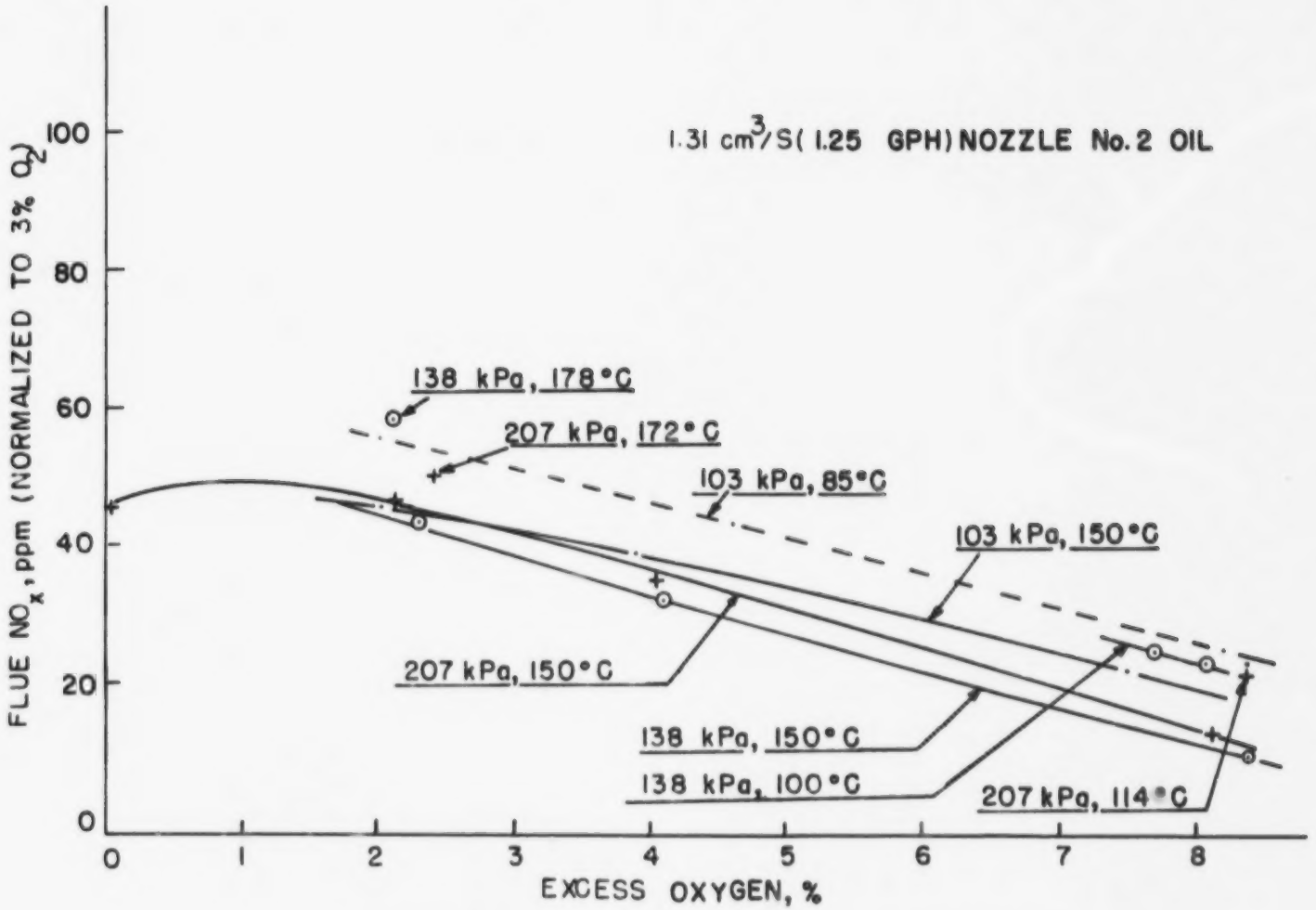

FIGURE 15. EFFECT OF OPERATING PARAMETERS ON NO_x

TRANSIENT PERFORMANCE

Some smoke, hydrocarbons, carbon monoxide and even methane were usually observed right after ignition. It seemed that these disappeared more quickly when hot fuel was used. This initial burst of emissions tended to saturate the sampling system and measuring instruments. Since the gas analysis system was slow to recover, we made a practice of removing the sampling tube from the combustion chamber during ignition. Although the instrumentation was not well suited to transient measurements, one transient experiment was conducted. The 0.85 gph nozzle was used with No. 2 oil at a pressure of 345 kPa(50 psi). Two runs were made; the first was with unheated oil and the second was with oil heated to yield a 150°C nozzle temperature. The results are presented in Table 5.

TABLE 5. EFFECT OF FUEL TEMPERATURE ON TRANSIENT PERFORM. 0.89 cm^3/s (0.85 GPH) NOZZLE, 345 kPa, No. 2 OIL

Time After Ignition Min.	Nozzle Temp. °C	Flue O_2 %	Smoke No.	NO_x PPM
0	35	---	---	---
1.5	80	---	5	---
5	88	---	0	---
10	100	3.1	0	40
15	100	3.0	6	40
20	100	3.0	5	41
0	71	---	---	---
1.5	121	---	0	---
5	150	5.0	1	26
10	150	4.7	0	28
15	150	4.5	0	28
20	150	4.5	3	29
23	150	4.5	1	29

After 1.5 minutes of operation the unheated oil gave a smoke No. 5 whereas the heated oil was already down to zero smoke. The nozzle with unheated oil rose to 80°C in 1.5 minutes whereas the nozzle temperature with heated oil reached 121°C in 1.5 minutes. Within 5 minutes the smoke No. was down to 0-1 in both tests. The excess oxygen was about 3% for the unheated oil case, whereas it was about 5% with heated oil. This difference in excess oxygen had no effect during steady state runs since smoke was unmeasurable unless excess oxygen was reduced well below 1%. For some unexplained reason, smoke increased again after 15 to 20 minutes. This phenomenon was not observed in any other tests which were run for as long as 30 minutes.

The NO_x level was higher with unheated oil. The increase was undoubtedly due to the lower excess oxygen. A knowledge of carbon monoxide and total hydrocarbon levels during ignition and shut down is needed to fully assess the effect of thermal aerosol atomization. However, response of the sampling train and measuring instruments was so slow compared to the combustion transients that these measurements were unreliable. Thus, the tests were not conclusive, but they did support the qualitative observation that heated fuel produced a shorter transient.

EFFECT OF SWIRL

The high swirl seemed to mask any effect of improved atomization with hot fuel. We therefore modified the burner to eliminate swirl. The objective was to try and answer the original hypothesis that improved atomization achieved through the use of hot fuel would reduce emissions and improve efficiency. Four additional air tubes were installed in the end of the mixing chamber to admit air parallel to the fuel jet axis. The tangential air jets were plugged. The results of a comparison using No. 1 oil and 1.05 cm^3/s (1 gph) nozzle are presented in Table 6. When the swirl was eliminated there was a correlation between smoke and fuel temperature. A nozzle fuel temperature of 114°C produced No. 9 smoke, 118 ppm CO, 5 ppm CH_4, 31 ppm total hydrocarbons and 11 ppm NO_x even with a excess oxygen of 4.9%. Increasing the fuel temperature to 179°C with essentially the same excess oxygen reduced all emissions except NO_x. A further increase in fuel temperature to 199°C and a decrease in excess oxygen to 4.1% reduced emissions still further. All emissions except NO_x disappeared when swirl was restored even though fuel temperature was varied from 121 to 178°C and excess oxygen was varied from 8.1 to 2.1%. Elimination of swirl appears to reduce NO_x slightly; however, this is achieved at increased levels of smoke, carbon monoxide and total hydrocarbons.

With swirl present the flame was very compact and non-luminous. Eliminating swirl produced a much larger volume luminous flame. The pictures in Figures 16 and 17 show this difference.

The total pressure of the air supplied when operating at 0.42 cm^3/s (0.4 gph) and 2% excess oxygen was about 1.2 kPa (5 in. water). This undoubtedly could be reduced by increasing the area of the air inlet passages and by reducing the swirl parameter to perhaps 2. Swirl does, however, add to the pressure requirement of the blower.

TABLE 6. EFFECT OF FUEL TEMPERATURE AND EXCESS OXYGEN ON SMOKE - No. 1 OIL, 1.05 cm^3/s (1 GPH) NOZZLE

Fuel Temp. °C	Fuel Press. kPa	Excess Oxygen %	Smoke No.	CO PPM	CH_4 PPM	THC PPM	NO_x
Without Swirl:							
114	138	4.9	9	118	5	31	11
179	138	4.8	3	45	0	5	23
199	138	4.1	4	10	0	3	19
With Swirl:							
121	138	8.1	0	0	0	0	18
163	138	5.1	1	0	0	0	25
159	138	5.4	0	0	0	0	29
178	138	2.1	0	0	0	0	63

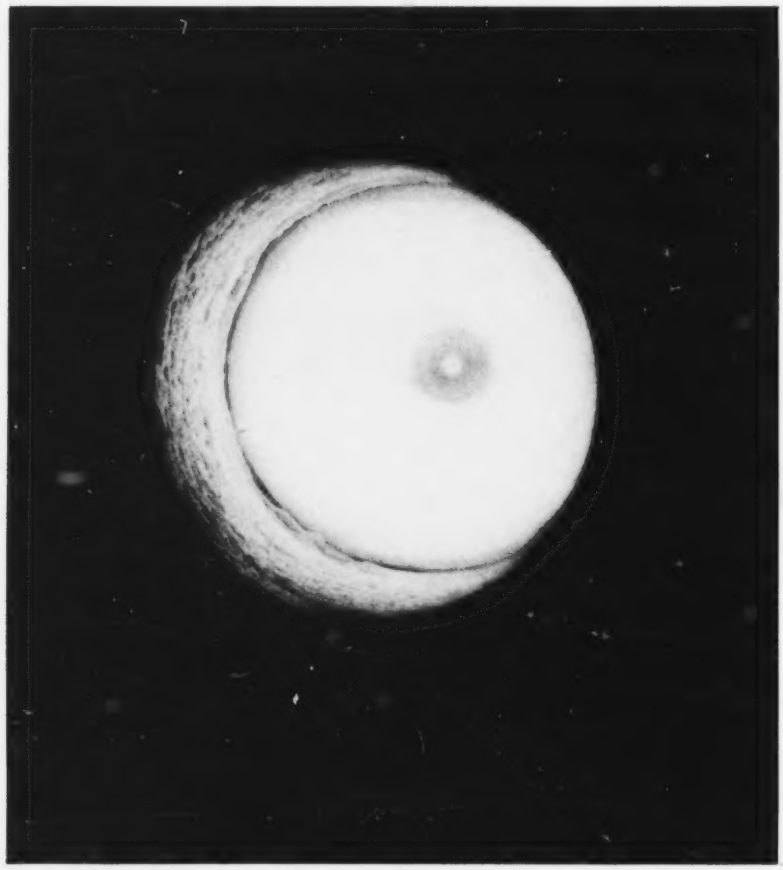

FIGURE 16. NON-LUMINOUS FLAME WITH SWIRL. DARK CENTER SPOT IS INLET ORIFICE WITH MIXING CHAMBER IN BACKGROUND.

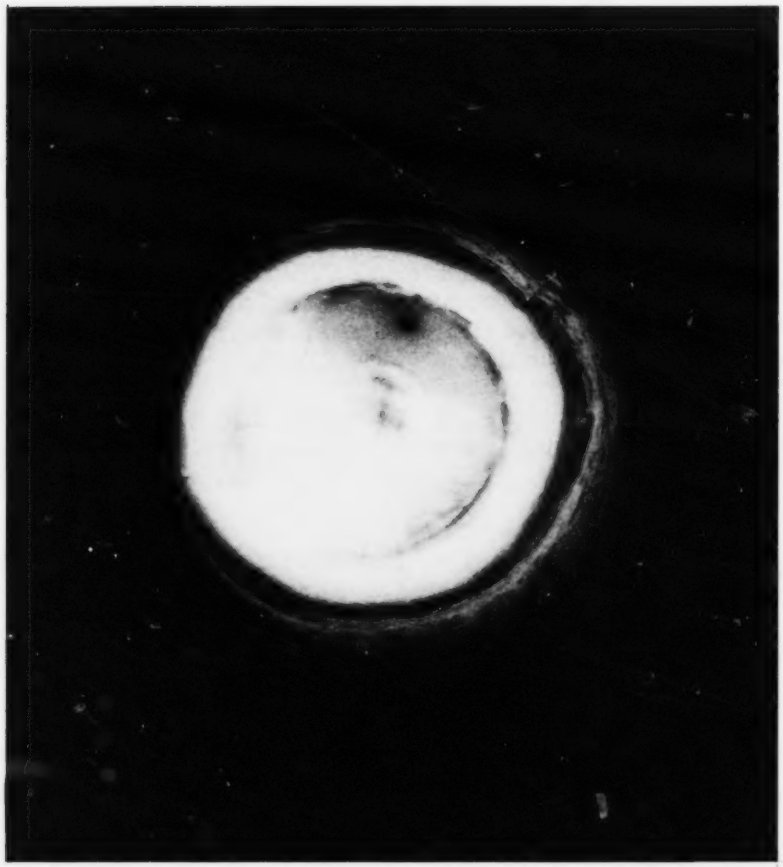

FIGURE 17. LUMINOUS FLAME WITHOUT SWIRL.

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APPENDIX

Drop Size

Surface tension is defined as the work required to displace a unit area of fluid surface.

$$\sigma = WK/A \tag{1}$$

For a spherical droplet,

$$WK = \sigma A = \sigma 4\pi r^2 \tag{2}$$

The volume of the droplet is,

$$V = (4/3)\,\pi r^3 \tag{3}$$

And the work per unit volume is,

$$WK/V = \frac{4\pi r^2}{(4/3)\,\pi r^3} = \frac{3\sigma}{r} \tag{4}$$

Thus the radius of the drop should be,

$$r = \frac{3\sigma}{WK/V} \tag{5}$$

Since work equals the change in energy,

$$r = \frac{3\sigma}{\Delta E/V} \tag{6}$$

Swirl Parameter

The swirl parameter is defined as the ratio of tangential velocity to axial velocity at the combustion chamber mixing orifice.

$$N_S = \frac{V_t}{V_a} \tag{7}$$

Air enters the vortex chamber tangential to the axis and at a radial distance, r_t. The air is accelerated due to the conservation of angular momentum as it goes through the mixing orifice which has a radius, r_o. Thus the tangential velocity at the orifice is:

$$V_t = \frac{Q\ r_t}{A_i\ r_o} \tag{8}$$

where: Q = air flow rate

A_i = area of air inlet tubes

r_t = radial distance of tubes from axis

r_o = radius of orifice

The axial velocity, V_a is

$$V_a = \frac{Q}{\pi\ r_o^2} \tag{9}$$

Then

$$N_S = \frac{Q\ r_t}{A_i\ r_o} \cdot \frac{\pi r_o^2}{Q} \tag{10}$$

Four the four air inlet tubes

$$A_i = 4\pi r_i^2$$

$$N_S = \frac{r_t\ r_o}{4\ r_i^2} \tag{11}$$

r_t = 1.5 in.

r_o = .75 in.

r_i = .25 in.

$$N_S = \frac{(1.5)\ (.75)}{4\ (.25)^2} = 4.5 \tag{12}$$